

Short communication

Thermoacoustically driven pulse tube cooler below 60 K

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Abstract

Modifications of hot-end heat exchanger, water cooler, hot buffer and acoustic pressure amplifier were carried out to improve the performance of a standing-wave thermoacoustic engine. A U-shaped pulse tube cooler was designed and fabricated according to the operating parameters of the engine. Experimental results of the pulse tube cooler driven by the modified thermoacoustic engine indicate that the pressure ratio has been increased from 1.152 to 1.179, and the cooling temperature has been dropped from 79.7 K to 56.4 K. © 2007 Elsevier Ltd. All rights reserved.

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1. Introduction

A pulse tube cooler driven by a thermoacoustic engine, without any moving components at both ambient and cryogenic temperatures, is a potential long-life cryocooler with high stability and reliability. Much progress has been made in recent years [1–7]. Tapered resonant tubes were introduced to thermoacoustic engines for suppressing harmonics and enhancing pressure oscillation in the engine [8,9], and an acoustic pressure amplifier was employed to increase pressure ratio [10], which is one of the key impacting factors for pulse tube refrigeration performance. A novel membrane configuration was used to separate the nitrogen-based thermoacoustic engine and helium-based pulse tube cooler [11], realizing higher pressure ratio and lower operating frequency. Cooling temperatures of 65.7 K and 41 K have been obtained by a single-stage and a two-stage pulse tube cooler, respectively, driven by a traveling-wave thermoacoustic engine [12,13]. A standing-wave thermoacoustically driven single-stage pulse tube cooler has achieved a cooling temperature as low as 79.7 K [14].

In order to enhance the driving performance, especially pressure ratio, modifications were focused on hot-end heat exchanger, water cooler, hot buffer and acoustic pressure amplifier of the standing-wave thermoacoustic engine reported in Ref. [14]. A U-shaped pulse tube cooler, which was expected to achieve a lower cooling temperature than that of previous coaxial one [14], was designed and fabricated according to the operating parameters of the engine. In the experiments of the U-shaped pulse tube cooler driven by the modified thermoacoustic engine, significantly improved performance has been observed.

2. Modifications of thermoacoustic system

The pulse tube cooler driven by the standing-wave thermoacoustic engine is schematically shown in Fig. 1.

The thermoacoustic engine employs a symmetric configuration, and consists of hot buffers, hot-end heat exchangers, stacks, water coolers and a resonant tube. Compared with the thermoacoustic system reported in Ref. [14], two new hot-end heat exchangers also with parallel plate channel were fabricated, and heating power capacity of each heat exchanger is up to 1.5 kW with porosity up to 0.33. In order to enhance heat transfer, the water coolers were changed from jacketed type to finned tube type. Figs. 2 and 3 are schematics of the hot-end heat exchanger and

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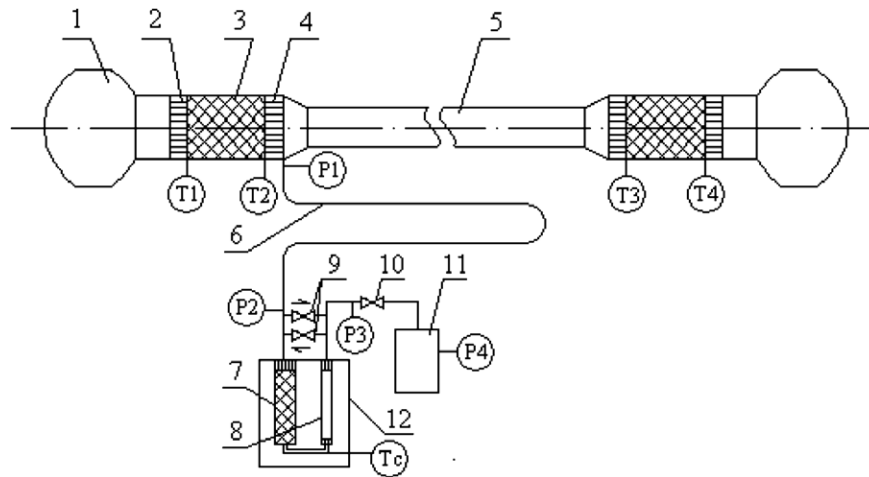


Fig. 1. Schematic of thermoacoustically driven pulse tube cooler; 1. hot buffer, 2. hot-end heat exchanger, 3. stack, 4. water cooler, 5. resonant tube, 6. acoustic pressure amplifier, 7. regenerator, 8. pulse tube, 9. second inlet, 10. orifice, 11. reservoir and 12. vacuum jacket.

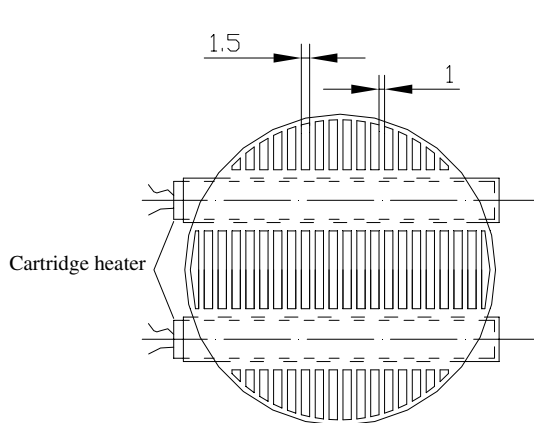


Fig. 2. Schematic of hot-end heat exchanger; ten cartridge heaters are inserted into a copper heat exchanger.

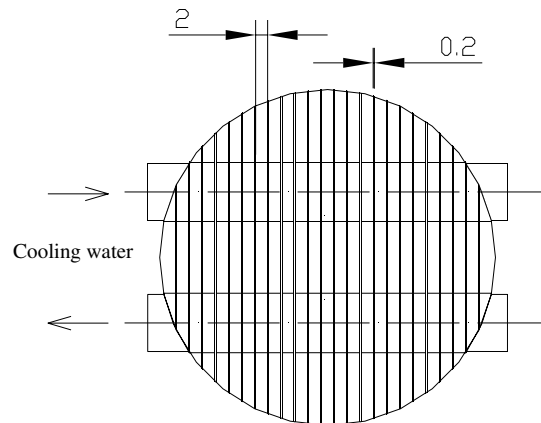


Fig. 3. Schematic of water cooler; aluminum fins are fixed out of copper tubes to enhance heat transfer.

the water cooler, respectively, and dimensions are tabulated in Table 1.

In addition, operating frequency transition from fundamental to second harmonic mode occurred in previous system [14], when the acoustic pressure amplifier was longer than 3.3 m. The volume of hot buffer was increased from 1.2 L to 1.35 L to suppress the frequency transition, and as a result, an acoustic pressure amplifier with length of 3.4 m worked successfully under fundamental frequency mode.

The U-shaped pulse tube cooler, as shown in Fig. 1, employed a double-inlet configuration. The second inlet consists of two needle valves in anti-parallel arrangement, and the DC flow circuiting in the loop composed of regen-

erator, pulse tube and second inlet can be effectively controlled by adjusting the dissymmetry of flow resistance in the second inlet. Dimensions of the cooler are presented in Table 2. The regenerator is packed with stainless steel screen disks of 400 mesh.

Table 2
Dimensions of pulse tube cooler

	Length (mm)	Outer diameter (mm)	Thickness of wall (mm)
Regenerator	57	12.5	0.2
Pulse tube	70	7	0.2

Table 1
Dimensions of hot-end heat exchanger and water cooler

	Length (mm)	Diameter (mm)	Flow channel		Fin thickness (mm)	Porosity
			Geometry	Space (mm)		
Hot-end heat exchanger	64	56	Parallel plate	1.5	1	0.33
Water cooler	34	56	Parallel plate	2	0.2	0.58 ^a

^a The porosity is calculated according to the section of minimum flow area.

Table 3
Operating parameters compared with data reported in Ref. [14]

Heating power (kW)	Length of acoustic pressure amplifier (m)	Heating temperature (K)	Frequency (Hz)	Mean working pressure (MPa)	Pressure amplitude (MPa)		Pressure ratio		No-load cooling temperature (K)
					P_1	P_2	P_1	P_2	
1.4	3.4	685.4	42.9	2.560	0.107	0.194	1.087	1.163	60.6
1.4	3.3	800.5	44.5	2.578	0.104	0.181	1.084	1.152	79.7

3. Experimental result and analysis

Experiment of the thermoacoustically driven pulse tube cooler was performed with helium as working gas. Measuring locations of temperature and pressure are also presented in Fig. 1. Heating temperatures T_1 and T_4 at the hot ends of stacks were measured by NiCr–NiSi thermocouples, while temperatures T_2 and T_3 at the cold ends of stacks by Cu–Constantan thermocouples. A Rh–Fe resistance thermometer was fixed at the cold tip of the pulse tube cooler to measure cooling temperature T_c . The pressures P_1 , P_2 , P_3 and P_4 were measured by four piezoresistive silicon pressure sensors.

Operating parameters are tabulated in Table 3, in which data from Ref. [14] are also listed for comparison. It is seen that with 1.4 kW heating power, pressure ratio at the warm end of regenerator was increased from 1.152 to 1.163, and no-load cooling temperature dropped from 79.7 K to 60.6 K. Thus, the above-mentioned modifications are effective. Additionally, heating temperature dropped markedly accompanied with a slight decrease of frequency, which is mainly derived from the enlarged hot buffer. The decreased heating temperature is helpful for the thermoacoustic system to make use of low grade heat energy.

Fig. 4 shows the variation of pressure ratio at warm end of the regenerator and cooling temperature with heating

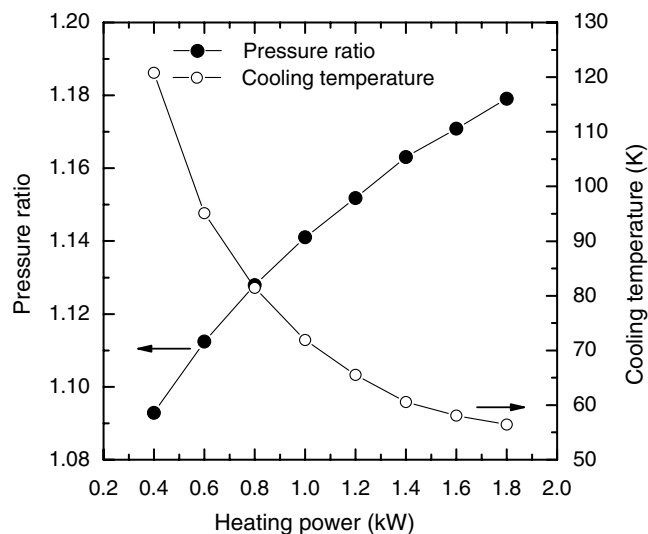


Fig. 4. Pressure ratio at warm end of regenerator and cooling temperature versus heating power.

power. It is seen that the pressure ratio rises with increase of heating power, and as a result, the cooling temperature decreases. A pressure ratio of 1.179 and a cooling temperature of 56.4 K have been achieved with heating power of 1.8 kW. Even with a heating power of 0.4 kW, the pressure ratio and cooling temperature can reach 1.093 and 120.1 K, respectively.

4. Conclusions

In comparison with the data reported in Ref. [14], the modifications of hot-end heat exchanger, water cooler, hot buffer, acoustic pressure amplifier and pulse tube cooler resulted in significant performance improvement for the thermoacoustically driven pulse tube cooler, including increased pressure ratio, lower cooling temperature and decreased heating temperature. The modified system has achieved a pressure ratio of 1.179 and a cooling temperature of 56.4 K, respectively, with heating power of 1.8 kW and heating temperature of 733.2 K.

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